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From Internal Sources to Urban Immission: Approaches to Understand Noise Propagation within Outdoor Heat Pumps and in Residential Areas

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Abstract

The sound of a heat pump, as perceived by owners and neighbors, results from an interaction between various internal sources (including the compressor, the cooling circuit flow, the ventilator fan, and electronics) and the transfer paths connecting these sources with the recipients. This paper presents the methods and findings of a research project that examines noise propagation from heat pumps from different levels of detail. In particular, the in-situ characterization of the compressor's blocked forces is described, alongside an analysis of how the placement and installation of heat pumps in residential settings alter perceived noise. Laboratory experiments, field measurements, and digital simulations, using sound power data and typical urban configuration models, demonstrate that measures such as optimized location, noise barriers, soundproof enclosures, and sound-absorbing facades can significantly reduce noise emissions. The study concludes with recommendations to improve the integration of noise mitigation strategies into urban planning.

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Keywords: Heat pumps; Compressor; Noise propagation; Noise reduction; residential areas; Simulation

1. Introduction

The sound of an air-source heat pump (ASHP), as perceived by owners and neighbors, originates from a complex interaction between internal sound sources, such as the compressor, the fan and the accompanying electronics, and the multiple transmission paths that convey the created sound by these sources to the listener. Thus, the noise one hears from an ASHP is determined not only by the intrinsic acoustical output of these internal sources but also by the structural transfer paths within the device and by the architectural and environmental pathways that shape the airborne sound propagation.

Not only is the directly radiated airborne sound from the internal sound source relevant, but also structure-borne sound or fluid-borne sound, which in turn can be radiated as airborne sound and thus become audible. A non-exhaustive overview of transfer paths and radiation mechanisms is provided in Figure 1. It becomes clear that the overall noise can be influenced by many factors. In this context, the compressor can be regarded as a source of airborne, structure-borne, and fluid-borne sound, which is why it is sensible to determine its intrinsic properties. Therefore, a method for determining the blocked forces as a measure of the intrinsic structure-borne sound characteristics of the compressor is considered in this paper.

In addition to source properties and device-level radiation and propagation, the residential environment also affects the perceived sound. The factors in Figure 1 illustrate these influences. This raises the question of how sound propagation can be simulated and which measures can positively affect it, leading to a reduction at the receiver point (RP).

Figure 1: Sound sources and propagation in ASHP and in a built environment. Based on [1],[2]

2. In-situ measurement of blocked forces

Understanding the noise emission due to vibrations of a ASHP starts with characterizing the compressor as a source of vibrations. An intrinsic measure of the structural excitation produced by the compressor are the blocked forces. Unlike the contact force (the actual forces acting at the connection points.), which depend on the properties of the mounting structure, the blocked force reflects the inherent vibratory output of the compressor independent from the mounting structure. This parameter is useful in understanding structure-borne sound sources and their contribution to the overall noise emission.

For its determination the indirect measurement based on the frequency response functions (FRF), as described in ISO 20270 [3], is the most practical approach for the application at ASHP and is therefore pursued further. In the frequency domain, the relationship is given by:

$$v_r(\omega) = Y_{rc}(\omega) \cdot f_c(\omega) \tag{1}$$

Where:

- $v_r(\omega)$ is the vector of measured velocity responses at the indicator DOFs (e.g. vibration measurements at the ASHP structure, refrigerant circuit or housing).
- $f_c(\omega)$ is the vector of blocked forces at the contact DOFs (e.g. the positions of the rubber feet). Contact DOFs refer to the degrees of freedom where a coupling between the source and receiver is assumed.
- $Y_{rc}(\omega)$ is the FRF matrix between indicator points "r" and contact points "c". Thus, translating how much vibrations are caused by the blocked forces at the indicator DOFs.

The blocked forces are then obtained by using the pseudoinverse $Y_{rc}(\omega)^+$.

$$f_c(\omega) = Y_{rc}(\omega)^+ \cdot v_r(\omega) \tag{2}$$

To verify the determined blocked forces validation DOFs are used. These are vibration measurements independent from the indicator DOFs, thus at different locations. Following the same concept as for the determination of the blocked forces. An FRF matrix $Y_{ic}(\omega)$ is needed to "connect" the blocked forces to the indicator DOFs ($v_i(\omega)$). A validation can then be made, by predicting the vibrations at the indicator DOFs by (3) and comparison to the measured vibrations.

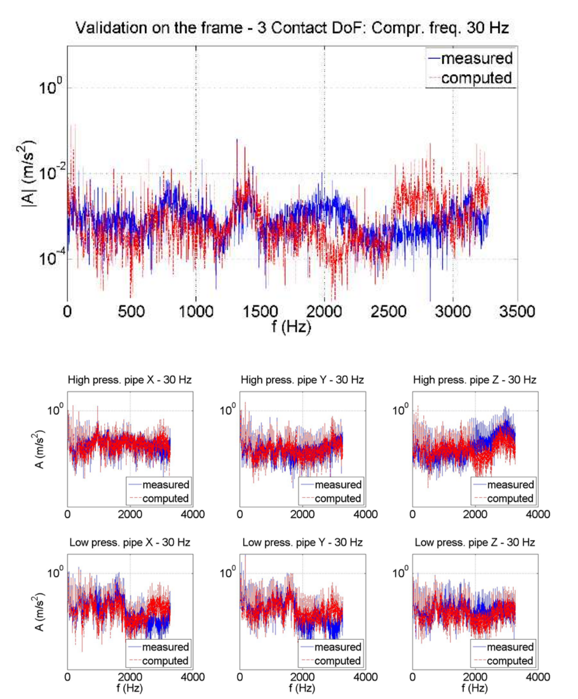
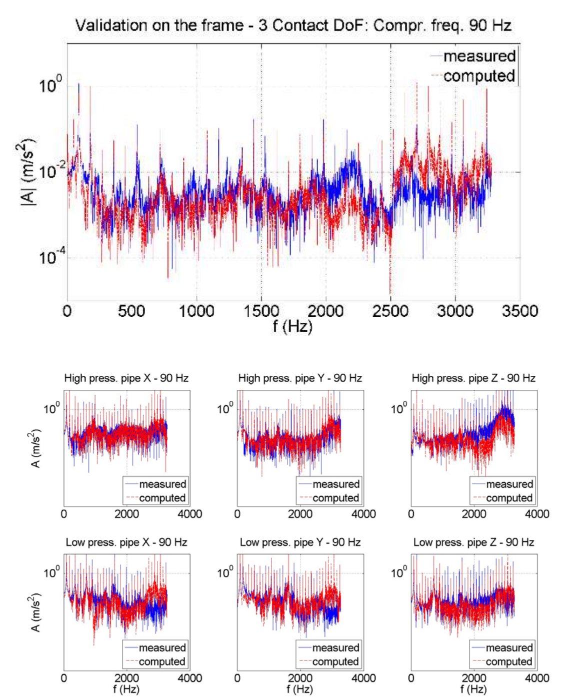
$$v_i(\omega) = Y_{ic}(\omega) \cdot f_c(\omega) \tag{3}$$

For the investigations a ASHP was used which features a compressor mounted on the three rubber feet. These rubber feet connect the compressor to the heat pump structure. Consequently, the interfaces of the rubber feet to mounting structure are considered as contact points. Initially the suction and discharge pipe connection points are considered as contact points as well. For validation vibrations are measured at the heat pump structure.

The determination of the blocked force depends on the assumed degrees of freedom (DOF) at the contact points. Three assumptions were considered. First, a 15 DOF approach considered blocked forces in all three spatial directions at both the rubber feet and the refrigerant circuit connection points (3 rubber feet × 3 DOF + 2 connection points × 3 DOF = 15 DOF). Second, a 9 DOF approach assumed that the dominant structural excitation occurs primarily through the rubber feet (3 rubber feet × 3 DOF), which improved numerical stability. Both approaches produced discrepancies between measured and predicted vibration responses at the validation DOF.

Third, a simplified 3 DOF model, assuming that the predominant vibration is in the vertical (z) direction, provides the most accurate prediction of the vibrations at the validation point. Additionally, the suction and discharge pipe DOFs are also considered as validation DOFs, also showing satisfactory deviations.

Table 1: Validation results for blocked force with 3 DOF

<table>
<tr><th>Validation at compressor speed 30 Hz</th><th>Validation at compressor speed 90 Hz</th></tr>
<tr><td>
</td><td>
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While a reduction in DOF (focusing on the z-direction) may neglect lateral effects, the validation results match the measurements the best. It can therefore be concluded that the vertical components of the blocked forces at the compressor feet are the most influential. This insight into the compressor's contribution via its blocked force lays the technical foundation for later evaluating the airborne noise emission that arises when the vibrations are transferred through heat pump and building structures.

3. Sound propagation in residential areas

The assessment of noise exposure from ASHP requires a robust description of the sound sources and of radiation and propagation processes up to the receiver. The aim of this study is to validate a simulation approach for sound propagation in residential settings by comparison with field measurements, enabling targeted evaluation of measures at the source with respect to their effect at the receiver. Input data comprise laboratory-derived, emission- and direction-dependent source characteristics and a geometrically sufficiently accurate representation of the built environment. Comparative simulations across four typical urban scenarios and three mitigation measures provide the basis for practical recommendations on estimating their noise-reduction effectiveness. The findings are relevant for planners and building owners and support the selection of appropriate measures.

3.1. Laboratory measurements

In the thermally insulated hemi-anechoic room at Fraunhofer IBP (Stuttgart), whose room-acoustic design meets DIN EN ISO 3745 Class 1 [4], the noise emission of an ASHP is characterized. The test facility was upgraded for measurements of airborne sound emissions in accordance with DIN EN 12102 [5]. ASHP are operated in heating mode at low temperature according to DIN EN ISO 14511-3 [6]. The thermal requirements for the operating points are as specified in [6], Table 12. The thermal boundary conditions are established by operating the unit under test. Up to three 3 kW fan heaters are available to stabilize the setpoint temperature. Directivity is measured using a semicircular microphone array: nine microphones at 20° azimuth spacing on a nominal 4.5 m semicircle (first position 10°, last 170°). The number and selection of measurement points follow ISO 26101-1 [7]. The heat pump is positioned centrally beneath the array. Directional scanning is performed by rotating the frame in 20° steps [8].

3.2. Validation measurements

The validation was based on structural and acoustic field data and on laboratory-derived source parameters. Measurements were conducted at a detached house on the Fraunhofer IBP campus in Holzkirchen. An ASHP was operated in front of the building. Sound levels were measured at three receiver points: at 5 m distance at 0° (①) and 90° (②), and at 0.5 m in front of a window located 2 m from the heat pump (③). Only microphone signals were evaluated. Predictions were performed with SoundPLANnoise®. The digital model was based on a 3D city model and was parameterized using measurements and plausible assumptions for source and surface properties, including reflection losses and ground factors. The heat pump was modelled as a point source at the device centre (see Figure 2, left). A reference point (④) coincident with the laboratory microphone position was included to test transferability. To emulate the hemi-anechoic field, the building was removed and the ground was set to acoustically hard. Levels were compared to the laboratory data, and the sound power was iteratively adjusted, if required. Calculated and measured levels showed small deviations. For a consistent spectral comparison (see Figure 2, right), the background noise measured in the field was added in the energy domain to the simulated receiver levels. Comparison with the laboratory situation confirmed the sound power and directivity. Minor fine-tuning reduced residual deviations. Overall, the model reproduces the field measurements with sufficient accuracy.

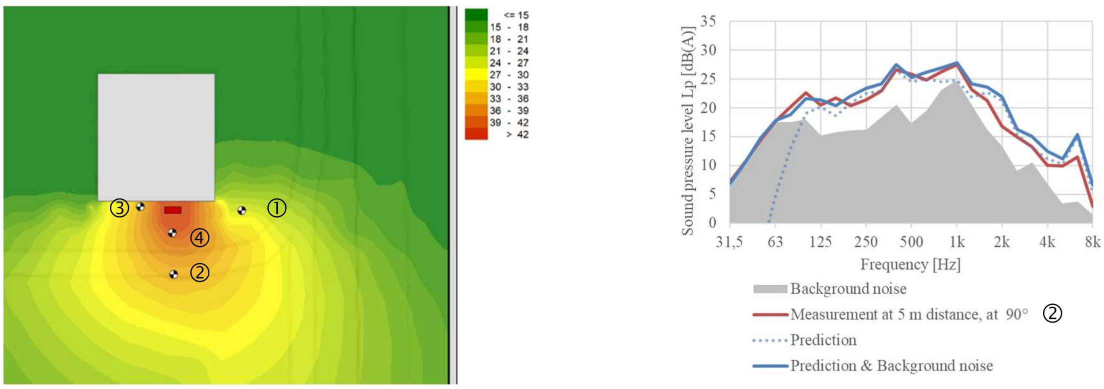

Figure 2: Left: noise map of the detached house with heat pump and emission points 1-4;
Right: Diagram showing the spectra of the measurement and the simulation prediction at a distance of 5 m and at 90°

3.3. Modelling of noise-mitigation measures in four residential areas

To assess noise reduction along the propagation path in residential areas, three mitigation measures were modelled and evaluated:

1. Sound-absorbing facade: 2 x 2 m, 50 mm mineral absorber placed in key reflection zones to reduce reflected components.
2. Barrier: rigid concrete structure 2.0 x 1.5 x 0.2 m, positioned 0.50–0.70 m in front of the heat pump on the source–receiver axis to block the line of sight.

3. Acoustic enclosure: sheet-metal casing with internal absorption and defined intake and exhaust openings, approx. 1.40 x 1.15 x 1.28 m.

The effectiveness of the noise-mitigation measures is assessed in four different residential environments. The assessment covers performance in the near field (⩽ 3 m), the far field, and at frequencies below 500 Hz. Three of the four residential environments are illustrated in Figure 3, while Figure 2 (left) illustrates the detached house.

Detached house: ASHP with three installation locations at the facade, in the front garden at greater distance, and on the flat roof. Receiver points are at the street-facing facade and around the corner to capture direct and screened positions. The barrier is placed along the source–receiver path. The enclosure is modelled either as a surface source or as a modified point source.

Terrace block: A long, two- to three-storey terrace block with two facade-near locations on one side. The distance to the opposite facade is approximately 25 m. Single and parallel operation allow analysis of additive effects. Receiver points on the opposite facade are arranged at different distances. A barrier in the principal propagation direction is included.

Fully enclosed greened courtyard: A fully enclosed greened courtyard 96 x 65 m with five- to six-storey perimeter blocks. Four source locations are considered, including the passageway, facade-near positions on both sides, and a balcony at 2.7 m height. Receiver points represent adjacent and opposite windows at various distances. Barriers are positioned relative to source and receiver. The enclosure fully encapsulates the source.

Enclosed courtyard: An enclosed courtyard 41 x 66 m. The courtyard plane is modelled as a roof to represent rooftop installations. Facade reflection loss is set to 1 dB. Installations in the courtyard and on the roof with one or two sources are considered. Noise immission at the receiving point is assessed within the courtyard. Measures are applied according to configuration.

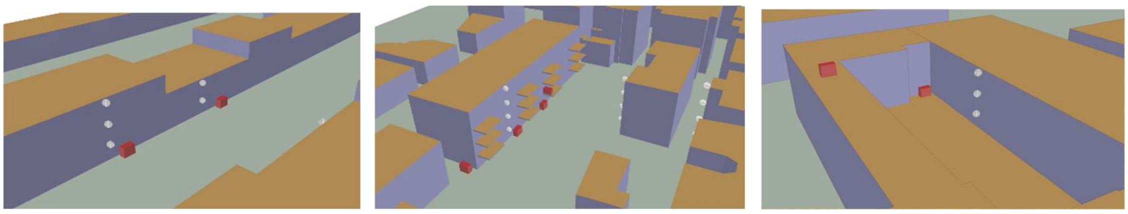

Figure 3: Three illustrations of the different residential environments, where the three noise-mitigation measures are applied, are shown. The red box indicates the position of the ASHP, and the white spheres indicate the position of the receiver point: left – Terrace block; middle – Fully enclosed greened courtyard; right – Enclosed courtyard

Across all scenarios, the acoustic enclosure showed the greatest effectiveness. Significant level reductions were achieved in both the near and far field, with far field reductions of up to 12 dB. Below 500 Hz, the effect was 1 - 4 dB lower. The barrier proved a robust alternative to the acoustic enclosure, with far field reductions of up to 11 dB and, where nearby receptor locations are directly screened, up to 8 dB. Effectiveness at frequencies below 500 Hz is reduced by up to 2 dB. Sound-absorbing facades produced no significant reductions at the assessed receptor locations (approximately 2 dB at most) and should therefore be considered as supplementary measures for specific situations (e.g. passageways).

Table 2. Overall conclusion of all noise-mitigation measures as a tabular summary

Measure	Reduction (near field)	Reduction (far field)	Effect below 500 Hz	Overall assessment
Acoustic enclosure	3 - 8 dB	9 - 12 dB	1 - 4 dB lower reduction	Effective at all points
Barrier	0 - 4 dB; up to 8 dB depending on siting	4 - 11 dB	Up to 2 dB lower reduction	Good alternative, especially for distant points or with direct screening of the RP
Sound-absorbing facade	≤ 2 dB	≤ 2 dB	No effect	Acoustically irrelevant; only useful as a supplementary measure
Siting of the ASHP	High leverage through smart placement	Case-dependent		Optimise early; can reduce the need for additional measures

Table 2 shows the overall conclusion of all noise-mitigation measures as a tabular summary. Overall, there is a clear emphasis on the acoustic enclosure as the primary measure, complemented by barriers, which provide consistent and reliable reductions and offer a cost-effective solution in many cases. Sound-absorbing facades should only be used selectively and in exceptional cases. Careful siting of the sound source proved to be a high-leverage measure and can further reduce the effort required for additional noise control measures. This comprehensive simulation approach, which combines laboratory emission characterization with detailed building geometry and subsequent field validation, forms a sound basis for evaluating and comparing noise reduction measures. It confirms that the simulation model is robust enough to predict the effect of various measures on the noise immission in a residential environment.

4. Conclusion

The noise characteristics of a heat pump is not only determined by the internal noise sources itself, but also by numerous transfer paths in the device and in the building environment. Thus, different methods are needed to understand these effects. Two aspects are discussed. First, it is shown, that the blocked forces can be determined by in-situ measurements for characterizing the structure-borne noise characteristics of the compressor. While the results dependent on the DOF considered, the method has proven to be practically feasible in ASHP. Second, an evaluation of sound propagation based on laboratory measurements, field studies, and numerical modelling shows that the building environment can influence noise immission at the receiver. The results show that combining targeted measures yields effective noise reduction. An acoustic enclosure offers the highest reduction potential, especially when complemented by noise barriers in areas where direct sound transmission is critical. These findings provide concrete recommendations for improving the installation and placement of heat pumps in urban and suburban areas, thereby enhancing overall noise compatibility.

Acknowledgements

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